Simulation of Torsional Vibration of Compressor Shafting

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Abstract: The dynamic model of the series complex rotor is established by focusing on the compressor shafting. Considering the excitation of the shafting frequency harmonics and the inter-harmonics of the motors, the rotor dynamics characteristics are calculated by means of transfer matrix method in DyRoBeS platform. The natural frequency and vibration mode of the shafting are obtained. And the shafting torsional vibration response of the harmonic excitation is analysed. In this paper, a simulation method for torsional vibration of complex shafting is formed, and the design principle of coupling for reducing the amplitude of torsional vibration of shafting is also proposed.

1. Introduction

The dynamic characteristics of the compressor shafting are related to the safety and stability of the equipment operation. It is important to analyse the rotor dynamics of the shafting to guide the safe operation of the equipment.

The compressor adopts the power form which two motors are synchronously driven, and the shafting structure is a typical long-axis system with multiple rotor series. The variable frequency speed control system is used to drive the shaft rotation. In general, the torsional vibration frequency of a single-span rotor driven by power frequency will be much higher than its working rotational speed, which is difficult to excite torsional vibration during operation. However, obvious torsional vibration of the variable speed driven compressor shafting sometimes will occur, which could destroy the shafting and endanger the safe operation of the equipment. Therefore, it is necessary to establish a detailed and accurate dynamic model of shafting and system excitation model, to analyse the torsional vibration characteristics of the shafting, and calculate the shafting response of the harmonic excitation of the motor, and to check the designing strength and rationality of the shafting and to improve the operation safety. At present, a variety of calculation methods have been developed for rotor dynamics, among which the transfer matrix method is one of the main methods to calculate the torsional vibration of the shaft system[1].

In this paper, the rotor dynamics model was established in view of the compressor shafting, the modal frequency, vibration pattern were calculated, torsional vibration steady state response of the shafting harmonic excitation and the inter- harmonic excitation of motors were analysed, and the design principle of coupling for reducing the amplitude of torsional vibration of shafting was proposed .

2. Dynamic Modelling of Shafting

The structure of the shafting is shown in Figure 1, using two motor synchronous to drive, through the intermediate shaft with diaphragm coupling connected to the two ends of the compressor. The dots on the axis represent a concentrated moment of inertia. Universally , the natural frequency of tangential vibration of long blades and ancillary structures of compressor are higher, which is not in the range of torsional vibration of shafting, simplify them as additional mass and the moment of inertia attached to the corresponding shaft segments when modelling[2]. The rotational damping of the dynamic pressure sliding bearing used in the system is relatively small, which has little effect on the torsional vibration of the shafting, and the empirical modal value is used in the calculation.

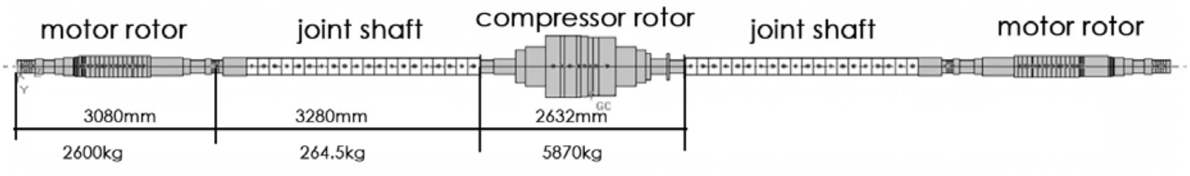


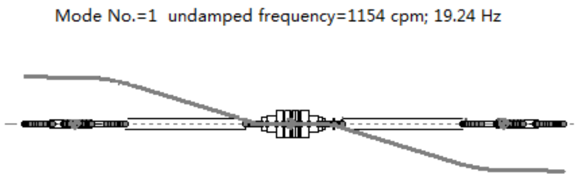
Figure 1 The layout of motor rotor and compressor rotor

The compressor rotor is connected to the motor rotor by a joint shaft with a diaphragm coupling, which is simplified to an equivalent optical axis for calculation [3]. The motor rotor is a complex rat cage structure, it is difficult to establish its precise numerical model completely, therefore using the recommendation of API624 [4], only consider the role of moment of inertia of the motor winding, ignoring its contribution to stiffness.

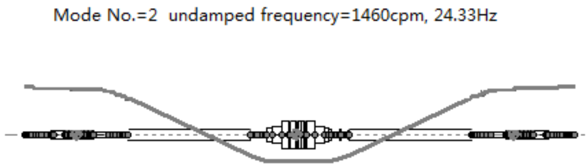
According to the above simplified method, the rotor dynamics model of the shafting is established in the rotor dynamics calculation software DyRoBeS, the torsional vibration characteristics of the shafting are analysed considering the excitation of the harmonic.

3. Modal Frequency and Mode of Torsional Vibration

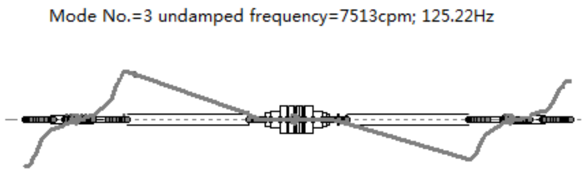
1154r/min, 1460 r/min, 7513 r/min, 7519 r/min and 9895 r/min are calculated for the first five orders unobstructed torsional modal frequencies of the shafting, and the vibration modes of each order torsional vibration frequencies are shown in Figure 2. The curve on the graph deviates from the axis displacement size to indicate the amplitude torsion angle of this shaft segment.



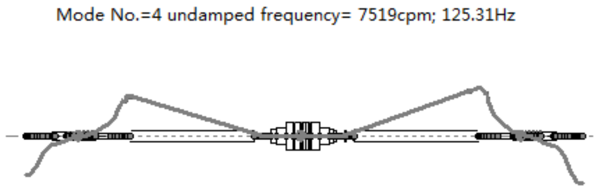
first-order torsional vibration mode



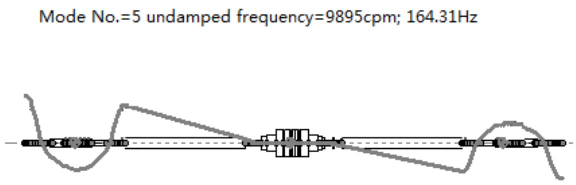
second-order torsional vibration mode



third-order torsional vibration mode



fourth-order torsional vibration mode



fifth-order torsional vibration mode

Figure 2 Rotor torsional vibration modes of shafting

The first-order mode is the joint shafts which are in the opposite phase torsional vibration. The second-order mode is the joint shafts which are in the same phase torsional vibration. The third-order mode is the motor rotor and the same side of the joint shaft are in the opposite phase torsional vibration and in the opposite phase of the other side. The fourth-order mode is the motor rotor and the same side of the joint shaft are in the opposite phase torsional vibration and in the same phase of the other side. In the fifth-order mode, significant torsional vibration occurs inside the motor rotor compared to other shaft segments. It can be discovered that the torsional vibration amplitude of the coupling parts are the largest in each order mode, which indicates that the couplings are the weakest segments in the shafting.

4. Excitation Response of Shafting

Figure 3 is the Campbell diagram of the rotor torsional vibration. The x direction is the rotor speed, while the y direction represents the frequency. The horizontal point drawing line represents the each order torsional mode frequency of the shafting, and the graph is marked with one time harmonic and two times harmonic of shafting and the inter-harmonic of the motor torque. Two vertical solid lines delineate the compressor operating speed range, for 1500rpm-3600rpm.

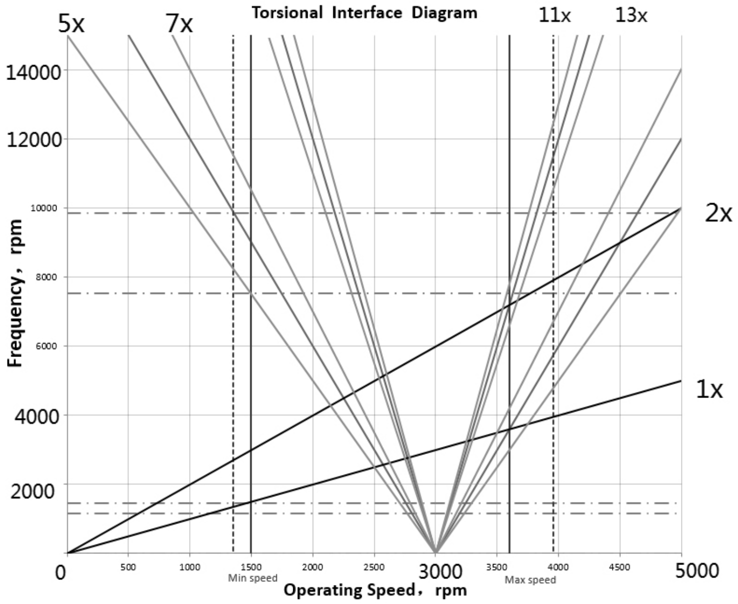


Figure 3 The Campbell diagram of the rotor torsional vibration

In the operating speed range, the rotor and motor excitation intersect with the first 5 order modal frequencies, which indicates that there is a risk of torsional vibration in the working speed range. It is necessary to analyse and evaluate the torsional resonance under the harmonic excitation of the shafting.

The harmonic of the excitation is more complex. Therefore, according to the practical experience from the project[5] and the data provided by the motor manufacturer, the harmonic excitation amplitude is determined as shown in Table 1.

Table 1 The amplitude of excitation

Type		Amplitude
Harmonics	1 X	1%
	2 X	1%
Inter-harmonics	5 X	0.02%
	7 X	0.14%
	11 X	0.1%
	13 X	0.06%

Under the excitation of 1X harmonic, the additional torsional stress respond of the motor rotor and the compressor rotor with the change of rotational speed as shown in Figure 4. At 1460rpm, the additional stress of the motor rotor peaks at 1.32MPa, and the additional torsional stress of the compressor rotor peaks at 1.58MPa, at which point the additional torque peak of the coupling is 1290Nm, as shown in Figure 5, and the torque response decreases rapidly after crossing the resonance point.

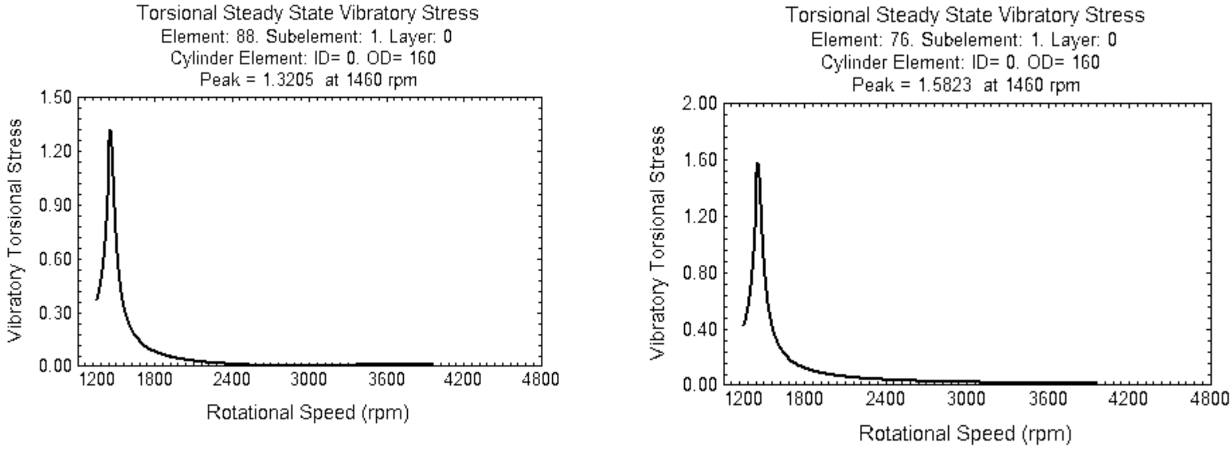


Figure 4 Torsional stress of motor rotor(left) and compressor rotor(right), 1X

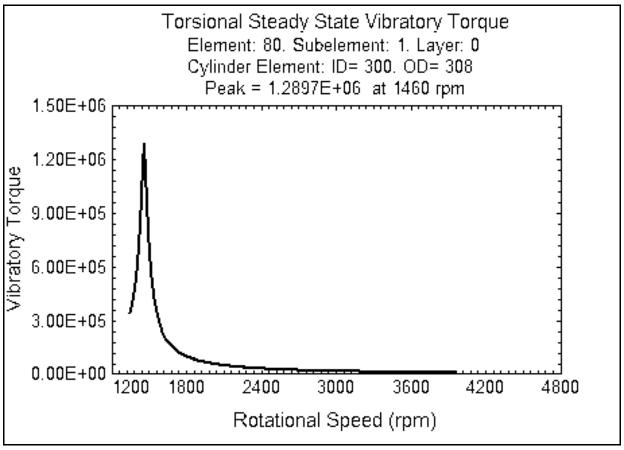


Figure 5 Torque response of coupling, 1X

By further calculation, the motor rotor, compressor rotor additional torsional stress and coupling peak additional torque with 2X harmonics and 5X, 7X, 11X and 13X inter-harmonic excitation are obtained, which are summarized as shown in Table 2.

Table 2 Peak torsional vibration response of shafting

location	Motor rotor/(MPa)	Compressor rotor/(MPa)	Coupling/(Nm)
1X	1.32 at 1460rpm	1.58 at 3750rpm	1290 at 1460rpm
2X	1.21 at 3750rpm	0.199 at 3750rpm	145 at 3760rpm
5X	0.025 at 2710rpm	0.03 at 2710rpm	24.6 at 2710rpm
7X	1.73 at 1590rpm	0.21 at 2790rpm	169 at 2790rpm
11X	0.29 at 2100rpm	0.11 at 2870rpm	91.6 at 2870rpm
13X	0.36 at 2240rpm	0.12 at 2910rpm	99.6 at 2910rpm

It can be observed that when the shaft speed is at 1460rpm, 1590rpm and 3750rpm, the motor shaft neck section, compressor shaft neck section will produce far more than the other speed range of high-frequency alternating stress, but the stress level of the steady state response on the shafting is at a low level, and it can be thought that the shafting has an infinite life[6].

It can be inferred that the diaphragm elastic coupling, which as the elastic part on the shafting, can effectively dissipate the torsional vibration energy of the motor frequency harmonic and other excitation inputs, reduce the amplification coefficient of the shafting under the harmonic excitation.

Therefore, reasonable design of the structural damping of the coupling could ensure the safe operation of the compressor shafting in the rated operating speed range.

5. Conclusion

In this paper, the torsional vibration characteristics of compressor shafting driven by frequency conversion are calculated by means of transfer matrix method in DyRoBeS platform.

The following conclusions are obtained:

(1) The series long shafting is a flexible rotor systems and the torsional vibration will occur under the harmonic excitation. Therefore, the torsional vibration level of the shafting needs to be checked when analysing the variable speed shafting.

(2) The diaphragm elastic coupling is the elastic part of the shafting. The structural damping of the coupling is reasonably designed, which can disperse the excitation energy of the system, inhibit the torsional vibration amplitude of the rotor and improve the safety of the equipment operation.

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